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Homogeneous charge compression ignition (HCCI) combustion: Mixture preparation and control strategies in diesel engines

Harisankar Bendu *, S. Murugan

Department of Mechanical Engineering, National Institute of Technology Rourkela, India

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ABSTRACT

At present achieving fuel economy and reducing emissions are the two main targets set by the automotive industries. Homogeneous charge compression ignition (HCCI) technology is believed to be a promising one to be applied in both spark ignition (SI) and compression ignition (CI) engines in the near future. However, some challenges such as compromise combustion phase control, controlled auto-ignition, operating range, homogeneous charge preparation, cold start and emissions of unburned hydro carbon (UHC), and carbon monoxide (CO) need to be overcome for successful operation of HCCI engine. Extensive research on HCCI combustion with a homogeneous fuel–air mixture preparation is going on throughout the world. This paper reviews the strategies of different external and in-cylinder mixture preparation methods which were adopted and proposed in the recent years. The different strategies of controlled auto-ignition by HCCI combustion are also discussed in this paper.

Contents

Abbreviations: ATAC, active thermal atmospheric combustion; ARC, activated radical combustion; CAD, crank angle degree; CAI, controlled auto-ignition; EGR, exhaust gas recirculation; HCCI, homogeneous charge compression ignition; HiMICS, homogeneous charge intelligent multiple injection combustion system; HTHR, high temperature heat release; OI, octane index; LTHR, low temperature heat release; LTC, LOw temperature combustion; MK, modulated kinetics; MULDIC, multiple stage diesel combustion; NADI™, narrow angle direct injection; PCI, premixed compression ignited; PFI, port fuel injection; PPCI, partially premixed compression ignition; PCCI, premixed charge compression ignition; PREDIC, premixed lean diesel combustion; SCCI, stratified charge compression ignition; UNIBUS, uniform bulky combustion system; VCR, variable compression ratio; VVA, variable valve actuation

* Corresponding author.
E-mail address: hsbendu@gmail.com (H. Bendu).

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1. Introduction

Internal combustion (IC) engines are the primary power horses in the automotive industries. Engines develop power by consuming a huge amount of fuel by combustion, and emit harmful exhaust emissions such as unburned hydrocarbon (UHC), carbon monoxide (CO), carbon dioxide (CO_2), oxides of nitrogen (NO_x), and particulate matter (PM) [1]. The automobile population is increasing exponentially due to the rapid growth in the population. The emission legislations also become stringent. The main task for the scientists, researchers, engineers and academicians is to find solutions to minimise the engine exhaust emissions, and effective utilisation of energy. Since last two decades, many automotive industries introduced several modern automotive vehicles, mainly to increase fuel economy, minimise the emissions, and to utilise different alternative fuels. In this regard, the researchers and engineers paid more attention towards the advanced modes of combustion like homogeneous charge compression ignition (HCCI), stratified charge compression ignition (SCCI), and low temperature combustion (LTC) due to superior thermal efficiencies and ultra-low emissions of NO_x and soot [1–5]. Among these, the HCCI engines have a potential to meet the stringent emission standards (EURO VI) and CO_2 emission standards [6,7]. Fig. 1 shows the region of HCCI combustion well above the UHC/CO oxidation limit and escapes the formation of both NO_x and soot. The HCCI combustion is considered to be one of the best combustion technologies to be adopted wider in the market near future, as it offers wide range of fuel flexibility [8–10] with a higher thermal efficiency, and low emissions. Fuel flexibility is also cited as a potential benefit [11]. The preparation of lean homogeneous mixture and low temperature combustion (LTC) are adopted in HCCI technology to suppress the NO_x and soot emissions from the engine [12].

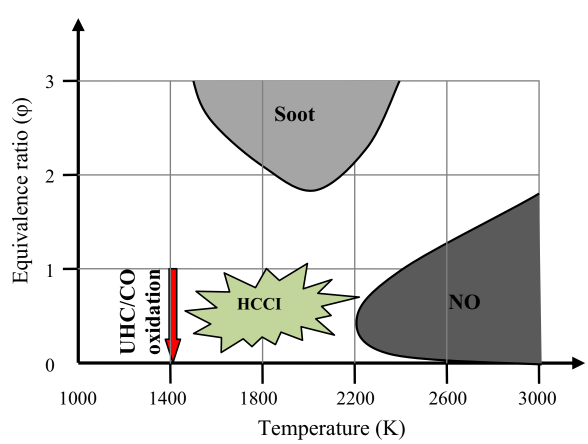

Fig. 1. HCCI combustion: simultaneous reduction of NO_x and soot.

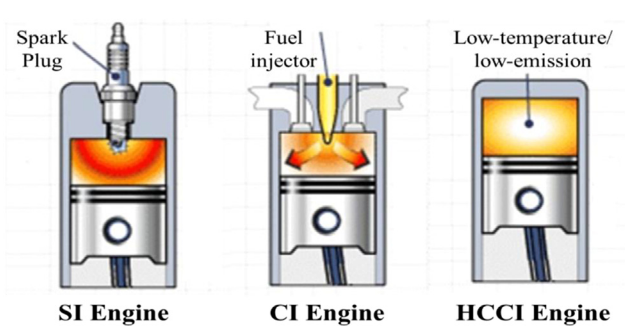

Fig. 2. Schematic diagram of HCCI [13].

Fig. 2 shows the comparison of SI, CI and HCCI operations. In HCCI engines, a lean homogeneous flammable mixture (fuel–air equivalence ratio $\Phi < 1$) is prepared, before the start of ignition and auto ignited as a consequence of temperature rise in the compression stroke.

The HCCI operation is alike to SI engine which utilises the homogeneous charge for combustion and alike to CI engine that has the auto ignition of the mixture. Thus, HCCI is the hybrid nature of SI and CI combustion processes [14]. In SI engines, three zones of combustion namely burnt zone, unburned zone and a thin flame reaction zone in-between for turbulent flame propagation through the cylinder. In CI engines, fuel is diffused into the cylinder and a definite diffusion flame travels with in the cylinder. In HCCI engine combustion spontaneous ignition of whole cylinder charge takes place without any diffusion flame or flame front propagation [15,16]. The comparison of different parameters influencing the combustion processes in SI, CI, and HCCI are given in Table 1.

Engines are operated in the region of lower equivalence ratios to improve efficiency and reduce emissions. Due to enormous increase in the vehicle population, the lean combustion technology is employed mainly in IC engines. The NO_x emission can be reduced only by reducing the flame temperature of combustion. Lean burn engines produce lower temperatures, which is the key factor to reduce the formation of thermal oxides of nitrogen. The excess air employed in lean burning results in a more complete combustion of the fuel which reduces both the hydrocarbon and carbon monoxide emissions. Moreover, the heat transfer losses in the IC engine can be decreased only minimising the combustion temperature. The HCCI combustion is one in which the low temperature combustion (LTC) is used to reduce the heat transfer losses, and the heat of fuel is completely released in a few crank angles near top dead centre (TDC).

The auto-ignition of the fuel by compression ignition (CI) engines can handle wide range of the fuels for combustion. The HCCI engines are operated to auto-ignite the fuel by the compression as the piston proceeds to the top dead centre. The engine has to be operated on a variable compression ratio (VCR) to adjust the auto-ignition of the cylinder charge near the TDC. A wide range of fuels can be burnt easily by adopting the VCR method [17–20]. Some of the other methods for fuel flexibility for the given engine are charge heating [21–23], boost pressure [20,24–29], exhaust gas recirculation (EGR) [30–32], variable valve actuation (VVA) [33–36] etc.

Table 1
Comparison of SI, CI and HCCI combustion engines.

Engine type	SI	HCCI	CI
Ignition method	Spark ignition	Auto-ignition	
Charge	Premixed homogeneous before ignition		In-cylinder heterogeneous
Ignition point	Single	Multiple	Single
Throttle loss	Yes	No	
Compression ratio	Low	High	
Speed	High	Low	
Combustion flame	Flame propagation	Multi-point auto-ignition	Diffusive flame
Fuel economy	Good	Best	Better
Max. efficiency	30%	> 40%	40%
Major emissions	HC, CO and NO_x	HC and CO	NO_x, PM and HC
Injection type	Port injection	Both port and direct injection	Direct injection
Equivalence ratio	1	< 1	
Fuel injection method	Direct injection Indirect injection		

2. Challenges of HCCI combustion

Before implementing the benefits of the HCCI combustion engines, it has to overcome some of the barriers for mass production. The challenges of the HCCI combustion include: (i) combustion phase control, (ii) controlled auto-ignition, (iii) operation range, (iv) cold start, (v) emissions of UHC and CO, and (vi) homogeneous charge preparation [37]. Among these challenges, the homogeneous mixture preparation and combustion phase control play vital role in determining the efficiency and emissions.

2.1. Combustion phase control

The main challenge of the HCCI engine is to control ignition timing, which influences the power and efficiency. The conventional engines have a direct mechanism to control the start of combustion. Unlike, spark timing in SI engines and fuel injection timing in CI engines, the HCCI engine lacks start of combustion controlled by auto-ignition. The fuel–air mixture is premixed homogeneously, before the start of combustion initiated by the auto-ignition of time–temperature history. This phenomenon of auto-ignition leads to the main combustion control which is affected by the few factors [37–39]: fuel auto-ignition chemistry and thermodynamic properties, combustion duration, wall temperatures, concentration of reacting species, residual rate, degree of mixture homogeneity, intake temperature, compression ratio, amount of EGR, engine speed, engine temperature, convective heat transfer to the engine, and other engine parameters. Hence, the HCCI combustion control over a wide range of speeds and loads is the most difficult task. Controlling combustion is the most important parameter, because it affects the power output and the engine efficiency. If combustion occurs too early, power drop in terms of efficiency and serious damage to the engine occurs, and if combustion occurs too late, the chance of misfire increases. Most of the researchers believe on the fact that HCCI combustion is governed by chemical kinetics [3,4,40,41].

2.2. Abnormal pressure rise with noise

The instantaneous heat release which is caused by auto-ignition of the whole homogeneous charge simultaneously during compression stroke. The instantaneous heat release results in abrupt rise in temperature followed by abrupt pressure rise, and then high levels of noise. Controlling this sudden heat release is extremely important, because it is the main source of pressure rise, which may cause a severe damage to the engine. The acceptable pressure rise limit is ≈ 8 bar/CA for noise [42].

2.3. Domain of operation

The operating range of HCCI engine is a limited one compared to the conventional engines, which is another hurdle for commercial success in the market. Controlling the ignition timing over a wide load and speed range is a difficult task [43,44]. The operating range is influenced mainly by the auto-ignition properties of fuel and engine geometry. Extending the HCCI operation to full/higher load also limits the part/light load operation, due to lack of ignition energy to auto-ignite the lean mixture at the end of compression stroke. The flammability limits the fuel–air mixture during very lean HCCI operation. In addition to this, the UHC and CO emissions also increase near the idle operation, as a result of inefficient combustion efficiency. Hence, the domain of operation of the HCCI engines is in limited range.

2.4. High levels of UHC and CO

The UHC and CO emissions in an IC engine are due to combustion of either rich or very lean to stoichiometric mixtures. The temperature of the lean mixture limits of inflammability, while the rich mixture suffers from lack of oxidant in the combustion chamber [45]. The efficiency of combustion is improved only, if the exhaust contains low levels of UHC emissions. The lean operation of the HCCI combustion produces high levels of UHC and CO emissions. The unburned hydrocarbon emission in HCCI engines is mainly due to the incomplete oxidation of fuel through an excess oxidant, which is available for combustion. Some other reasons of UHC are crevice volumes present in the combustion chamber, valve overlapping, wall deposit absorption etc [46]. The exhaust gas temperature is too low to oxidise UHC and CO to CO_2 and H_2O completely, even during exhaust stroke. Due to low temperature combustion process, catalytic converters are also inefficient to oxidise these pollutants. The efficiency of combustion in HCCI is greatly improved by the complete oxidation of the fuel lastly by power stroke.

2.5. Cold start

The cold start problem in the HCCI engines is another hurdle in most of the geographically cold regions. The compressed charge looses more heat to the cold combustion chamber walls at the cold start operation. This problem can be overcome by starting the engine by the conventional mode for a short warm-up period, and then switch to the HCCI mode.

2.6. Homogeneous charge preparation

The mixture preparation is the key to achieve high fuel economy and low exhaust emissions from the engine. The thermodynamic cycle time of internal combustion engines takes a very short span and within that, the homogeneous mixture preparation time for combustion is much lower. The degree of homogeneity of the fuel–air mixture is greatly improved only by increasing the time for mixture preparation. Some other benefits of effective mixture preparation are control of wall wetting and oil dilution. LTC was employed in many combustors like IC engines, and gas turbines mainly to decrease NO_x emissions as they are responsible for peroxyacetyl nitrates (PAN) formation. The classification of LTC for homogeneous charge preparation strategies which are implemented in IC engines are shown in Fig. 3.

3. Homogeneous charge preparation strategies

The preparation of the homogeneous mixture is the main factor in reducing the particulate matter (soot) emissions, and local fuel rich regions to minimise oxides of nitrogen. The local fuel-rich regions can be decreased by an effective mixture preparation.

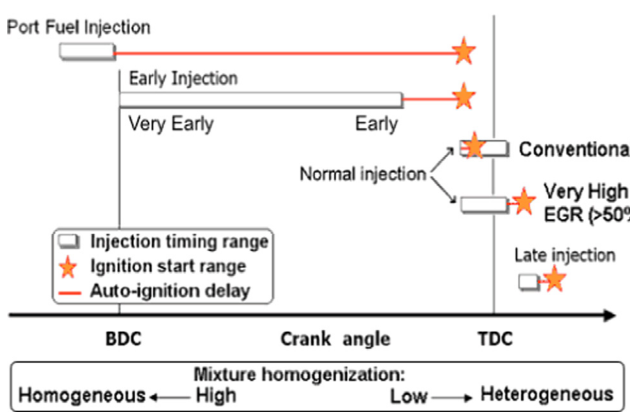

Fig. 3. Classification of low-temperature combustion strategies [47].

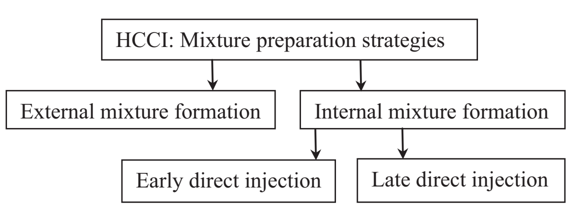

Fig. 4. Strategies for mixture preparation.

However, the preparation of the homogeneous mixture for the cycle-to-cycle variation of speed and load is a difficult task due to less time availability of mixture preparation. The effective mixture preparation for the HCCI combustion includes both the fuel–air homogenisation and temperature control over in combustion. The strategies for mixture preparation are either in-cylinder direct injection, or external mixture which is shown in Fig. 4. Both the preparation methods have their own disadvantages that the external mixture has a low volumetric efficiency and in-cylinder mixture is prone to an oil dilution. Table 2 gives the various names of HCCI which are listed in the literature. This session describes the strategies and implementations of mixture preparation.

3.1. External mixture preparation

The homogeneous mixture which is prepared external to the engine cylinder is the most effective due to more mixing time availability, before the start of combustion. This method of preparation is more suitable for high volatile fuels like gasoline and alcohols. The mixture preparation strategies are port fuel injection (PFI), manifold induction, fumigation, wide open throttle (WOT) carburetion etc. However, the low volatile fuel like diesel can also be used by using fuel vaporiser. The gaseous fuels are ready to mix with the air and preparation of homogeneous mixture externally is pretty simple, but the engine may suffer with the volumetric efficiency, if the calorific value of the gas is low. The gaseous fuels are mixed mostly in the intake manifold and some early implementations are acetylene [21,48], biogas [49–51], hydrogen [52–55] etc. Fig. 5 illustrates the different methods of external mixture preparation by researchers.

A first study on HCCI combustion process has been performed on two stroke engines by Onishi et al. in 1979 [15]. There is no flame propagation, as in a conventional SI engine instead, the whole mixture burns slowly at the same time. They called it active-thermo atmosphere combustion (ATAC) [56–58]. The same combustion was demonstrated at the Toyata Motor Co. Ltd. and named as "Toyota-Soken (TS) combustion" [16]. Noguchi et al. [16] demonstrated the same combustion process in an opposed-piston two stroke engine. Later, Honda R&D Co., Ltd. investigated on

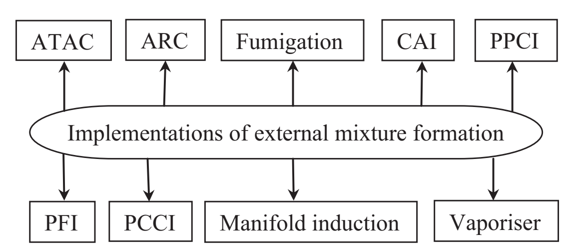

Fig. 5. Early implementations of external mixture formation.

Table 2
Overview of HCCI acronyms from literature.

Reference	Acronym	Meaning	Location
Onishi [15]	ATAC	Active thermo-atmosphere combustion	Nippon Clean Engine Research Institute
Noguchi [16]	TS	Toyota-Soken combustion	Toyota/Soken
Thring [153]	HCCI	Homogeneous charge compression ignition	Southwest Research Institute (SwRI)
Ishibashi [59]	ARC	Active radical combustion	Honda
Gatellier [106]	NADI™	Narrow Angle Direct Injection	Institut Français Du Pétrole (IFP)
Kimura [100]	MK combustion	Modulated kinetics combustion	Nissan
Takeda [91]	PREDIC	Premixed diesel combustion	New ACE
Hashizume [94]	MULDIC	Multiple stage diesel combustion	New ACE
Yokota [95]	HiMICS	Homogeneous charge intelligent multiple injection combustion system	Hino
Yanagihara [96]	UNIBUS	Uniform bulky combustion system	Toyota
Iwabuchi [93]	PCI	Premixed compression ignited combustion	Mitsubishi
Aoyama [65]	PCCI	Premixed Charge Compression Ignition	Toyota

activated radical combustion (ARC) on two stroke gasoline engines [59–63] by winning the fifth place in Granada-Dakar rally competition. During radical combustion, the exhaust port throttling has been tested at a range of 2 to 16 mm (exhaust port reduction area is 1–8%) by Saqaff et. al. [64] on a two stroke engine. It is reported that, the exhaust gas temperature decreased by about 16.7–22.5% at all engine speeds and loads, while the fuel consumption reduced by about 11.1–49.8%. The PCCI (premixed charge compression ignition) engine developed by the Toyota Central Research [65] in which combustion of premixed lean mixture arises from a multi-point ignition is very promising and necessary for achieving both higher efficiency and lower nitrogen oxide (NO_x) emission. The PCCI engine operates stably in the air–fuel ratio range of 33–44 and ignition occurs spontaneously at unspecified points as it does in diesel engines. Table 3 shows the external the mixture preparation strategies used in HCCI engines.

Some researchers introduced an electronically controlled fuel vaporiser for low volatile and high boiling point fuel such as diesel [66–69]. The diesel vaporiser formed a very light and dispersed aerosol with a very fast evaporation due to a very high surface to volume ratio. The smoke emissions were reported to be negligible and the EGR was used for combustion control and the NO_x emissions. The operation temperature of vaporiser is above the boiling point of fuel for successful external mixture preparation [70]. Some researchers used a high intake air temperature [71–75] to vaporise the fuel in the intake manifold. The common disadvantage reported by them is the electric power consumption for vaporisation of diesel. Another study was reported by the researchers on the effect of premixed ratio in diesel engine with the partially premixed charge compression ignition (PPCI) combustion using diesel fuel [76–78]. An investigation [17] of diesel fuel with a port fuel injection with variable compression ratios reported that, the compression ratios need to be reduced in order to avoid knock in the HCCI combustion. The cool-combustion chemistry of diesel fuel leads to auto-ignition at approximately 800 K during compression stroke [79].

The port fuel injection (PFI) is the simplest method of external mixture preparation, in which injector is mounted in the intake manifold, very close to the intake valve. This system improves the volumetric efficiency and fuel distribution over carburetion. The mixture enters into the cylinder during engine suction and the turbulence created by intake flow improves further homogenisation. This method of mixture formation has been reported to be successful with gasoline and alcoholic fuels [80–87]. The main drawback of this strategy is injection timing cannot influence the start of ignition. Furthermore, heavy fuels with lower volatility of PFI results in poor vaporisation with increased wall impingements.

3.2. In-cylinder mixture preparation

The demerits associated with diesel-fueled by the port fuel injection with an internal mixture formation has been investigated. Two strategies: (i) early direct injection and (ii) late direct injection for in-cylinder mixture formation have been adopted in the study. The injection timing for early direct injection was set during compression stroke, for late direct injection it was set after TDC. High injection pressures with a large number of small nozzle holes adopted to increase the spray disintegration which forms homogeneous mixture.

3.2.1. Early direct injection

The fuel injection process in the HCCI combustion is charge homogeneity, which is influenced by injection timing. Early injection method is mostly used method of achieving HCCI diesel combustion. The early injection allows a longer ignition delay along with the low temperatures to homogenise the diesel–air mixture. Unlike conventional diesel, direct injection in diesel engines, pulsed injection strategy is used. The total amount of fuel per cycle is injected in many pulses as shown in Fig. 6.

The early direct fuel injection during compression stroke results in wall wetting due to over-penetration of diesel as a result of poor volatility and low air density during early CAD compression stroke. A piezoelectric controlled common rail injector is capable to control injection with high injection pressures for

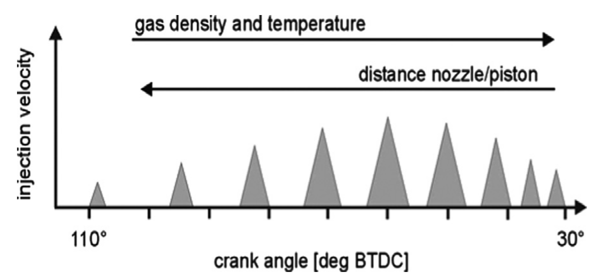

Fig. 6. Pulsed injection strategy for early in-cylinder injection [88].

Table 3
Overview of external mixture preparation strategy implemented in gasoline-fueled HCCI engines.

Reference[a]	HCCI acronym	Key features	Advantages	Disadvantages
Onishi [15]	ATAC	Uniform mixing between residuals and fresh charge. No flame propagation as in the case of SI engines. High EGR rates are used to achieve auto-ignition of gasoline.	Remarkable reduction in emissions and high fuel efficiency.	Limited to part load operation.
Noguchi [16]	TS	Stable spontaneous auto-ignition with port fuelling in presence of active radicals.	Smooth combustion with low HC emissions and improved fuel consumption.	Limited to part load operation.
Thring [153]	HCCI	The operating regime was function of air/fuel equivalence ratio and external EGR rates.	High fuel efficiency and low emissions.	Restricted to part load operation and control of auto-ignition timing is problematic.
Ishibashi [59]	ARC	Active radicals in the exhaust gases were controlled by changing the exhaust valve axis movement.	Two-stage auto-ignition combustion is observed at lower load. Fuel economy was improved by 57% while HC emission reduction by 60%.	Idling with auto-ignition was not possible with AR combustion.
Saqaff [64]	ARC	Reduced exhaust port area in the range of 1–8%.	Exhaust gas temperature decreased by about 16.7% to 22.5% while the fuel consumption reduced by about 11.1% to 49.8%.	Idling with auto-ignition was not possible with AR combustion.
Aoyama [65]	PCCI	Spontaneous ignition occurred at unspecified points as it does in diesel engines. The flame then developed rapidly throughout the combustion chamber.	Low NO_x emission was noticed than in diesel engines.	Intake air heating and supercharging were necessary to extend the range of stable combustion.

[a] Only the first author is listed, all papers have multiple authors.

Table 4
Overview of in-cylinder direct injection strategies used in diesel-fueled HCCI engines.

Reference[a]	Strategy	HCCI acronym	Key features	Advantages	Disadvantages
Takeda [91]	Early in-cylinder direct injection	PREDIC	Two side injectors whose nozzle diameter was reduced from 0.17 mm to 0.08 mm and the number of holes increased from 6 to 16.	Low NO_x and soot emissions.	The UHC emission was more due to poor combustion efficiency. Limited to part load operation only.
Nishijima [92]	Early in-cylinder direct injection	PREDIC	Two sprays, one injected from each side injector diagonally, impinged each other at the centre of the cylinder to create an air–fuel mixture.	Adequate water split injection pattern could retard fuel injection timing by keeping low NO_x and UHC emission levels.	Due to wall impingement of the fuel spray, higher UHC emission and fuel consumption were observed.
Iwabuchi [93]	Early in-cylinder direct injection	PCI	Used an impinged-spray nozzle which realized the low penetration, high-dispersion, and high injection rates.	NO_x emissions reduced.	Limited to part load operation and found more UHC emissions.
Hashizume [94]	Early in-cylinder direct injection	MULDIC	Two stage injections: the first stage combustion corresponds to the premixed lean combustion, and the second stage combustion corresponded to diffusion combustion under the high temperature and low oxygen conditions.	Reduce NO_x emissions at high load conditions.	UHC emissions were higher.
Yokota [95]	Early in-cylinder direct injection	HiMICS	Two injector tips were needed, a multi-hole (30 holes each 0.10 mm) tip for early injection and a conventional injector tip for near TDC injection.	NO_x reduction from approximately 800 to 200 ppm.	UHC increased from 3000 to 8000 ppm.
Hasegawa [96]	Early in-cylinder direct injection	UNIBUS	A double injection technique was used. The first injection was used as an early injection for fuel diffusion whereas the second injection was used as an ignition trigger for all the fuel.	Low NO_x and smoke emissions were possible.	Limited to low loads and at high loads conventional diesel combustion was used.
Su [97,98]	Early in-cylinder direct injection	MULINBUMP	A flash mixing technology was used with BUMP combustion chamber, while multi-pulse injections to limit the spray penetration.	Near zero NO_x and smoke emissions.	For higher power output injection mode must be carefully designed.
Kimura [100]	Late direct injection	MK combustion	Three features in MK combustion: Late fuel injection timing starts from 7° BTDC to 3° ATDC, high levels of EGR and high swirl ratio.	NO_x emissions were reduced to about 50 ppm without an increase in PM.	The operating range was limited to one-third of peak torque and half-speed.
Kimura [101]	Late direct injection	MK combustion	In addition to the above, piston bowl diameter was increased from 47 to 56 mm. This reduced UHC emissions significantly.	ultra-low emission vehicle (ULEV) standards can be met with a 5-way catalyst. $NO_x < 0.07$ g/mile and $PM < 0.01$ g/mile.	The operating range was limited to half torque and three-quarters speed.
Gatellier [105–107]	Early in-cylinder direct injection	NADI™	The fuel injection angle was narrowed such that wall wetting was eliminated.	At part load near zero particulate and the NO_x emissions while maintaining very good fuel efficiency.	Conventional diesel combustion was observed at full load.

[a] Only the first author is listed, all papers have multiple authors.

performing variable pulsed injections. The area below the curve represents the fuel mass belonging to each pulse. The low gas density at the beginning of injection requires short pulses with the reduced injection velocities, and the time interval between the pulses is relatively large. As the piston moves up, density and temperature in the cylinder increase and penetration is reduced. The pulse durations can be prolonged, while the time intervals between subsequent pulses are decreased. At the end of the pulsed injection the distance between nozzle and piston reduces significantly, and the mass injected per pulse must be reduced again in order to prevent fuel deposition on the piston [88]. The in-cylinder mixture preparation strategies used in HCCI engines are listed in Table 4.

The early in-cylinder implementations used in diesel-fueled vehicles are PREDIC, MULDIC, HiMICS, UNIBUS and MULINBUMP. Fig. 7 illustrates different direct in-cylinder strategies adopted in recent years.

The fuel is injected during early compression stroke, which becomes partly homogeneous mixture and combustion starts closer to the TDC. This concept is called premixed lean diesel combustion (PREDIC) [89,90]. Takeda et al. [91] from New ACE institute, Japan reported low NO_x and soot emissions, while the UHC emission is more due to poor combustion efficiency. The injection strategy was modified such that, two side injectors whose nozzle diameter was reduced from 0.17 mm to 0.08 mm

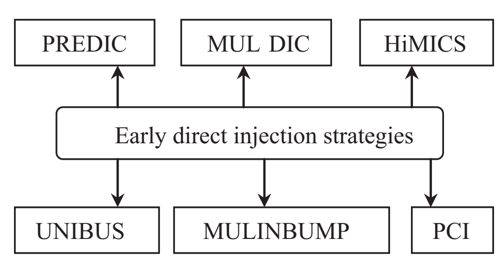

Fig. 7. Early in-cylinder diesel direct injection strategies.

and the number of holes was also increased from 6 to 16. The operating region is limited to part load only. Nishijima et al. [92] used an early injection timing and found a problem that the fuel spray reaches the cylinder wall, which causes a higher HC emission and fuel consumption. Iwabuchi et al. [93] from the Mitsubishi Motors corporation used early injection strategy, where premixed compression ignited (PCI) combustion system adopted in a four stroke, single cylinder, diesel engine. The PCI combustion limited to part load and found more HC emissions with low NO_x emissions.

The strategy of multiple injections in the HCCI mode is used instead of a single injection strategy, in order to run at high load. The multiple stage diesel combustion (MULDIC) and homogeneous charge intelligent multiple injection combustion system (HiMICS)

uses the first injection during early compression stroke and the second injection is just before the TDC. Hashizume et al. [94] studied the MULDIC in which the first stage combustion corresponded to the premixed lean combustion and the second stage combustion corresponds to diffusion combustion under the high temperature and low oxygen conditions. The HiMICS concept based on pre-mixed compression ignition combustion combined with a multiple injection developed by the Hino Motors, Ltd. [95]. The pre-mixture was formed by a preliminary injection performed during a period from the early stage of the induction stroke to the middle stage of the compression stroke and later injection after TDC to oxidise soot. The emissions of NO_x and soot are reported less, but high levels of HC and CO.

The uniform bulky combustion system (UNIBUS) developed by the Toyota Motor Corporation in the Japanese market (1KD-FTV, 3 l-4cylinder) in August 2000 [96]. A double injection technique was used. The first injection was used as an early injection for fuel diffusion and to advance the changing of fuel to lower hydrocarbons (i.e. low temperature reaction). The second injection was used as an ignition trigger for all the fuel. It is reported, that the ignition of the premixed gas could be controlled by the second injection, when the early injection was maintaining a low temperature reaction. The low NO_x and smoke emissions are possible both in the first injection and in the second injection by this combustion. This system is limited to low loads and at high loads conventional diesel combustion is used.

The MULINBUMP is a compound combustion technology of premixed combustion and “lean diffusion combustion” in a DI diesel engine [97]. The premixed combustion is achieved by the technology of multi-pulse fuel injection. The start of pulse injection, injection-pulse number, injection period of each pulse and the dwell time between the injection pulses are controlled. The objective of controlling the pulse injection is to limit the spray penetration of the pulse injection, so that the fuel will not impinge on the cylinder liner, and to enhance the mixing rate of each fuel parcel by promoting the disturbance to the fuel parcels. The last or main injection pulse was set around the TDC. A flash mixing technology was developed from the design of a so-called BUMP combustion chamber, which was designed with some special bump rings. The combustion of fuel injected in the main injection proceeds under the effect of the BUMP combustion chamber at a much higher air/fuel mixing rate than it does in a conventional DI diesel engine, which leads to “lean diffusion combustion”. The pulse injection mode modulation was investigated by variation of control signals, a series of injection modes were realized based on the prejudgment of combustion requirement. The designed injection modes included, so called even mode, staggered mode, hump mode and progressive increase mode with four, five and six pulses respectively. An engine test was conducted with the designed injection modes. The experimental results showed that the HCCI diesel combustion was extremely sensitive to the injection mode. There were many ways to reach near zero NO_x and smoke emissions, but the injection mode must be carefully designed for higher power output [98].

3.2.2. Late direct injection

The development of diesel-fueled late DI HCCI system is the modulated kinetics (MK) combustion system developed by the Nissan Motor Co., Ltd. [99,100]. A schematic diagram of the Nissan MK-concept is shown in Fig. 8 [101]. This system combines two mutually independent intake ports, one of which is a helical port for generating an ultra-high swirl ratio and the other is a tangential port for generating a low swirl ratio. The tangential port incorporates a swirl control valve that controls the swirl ratio (3.5–10) by varying the flow rate. To achieve the premixed

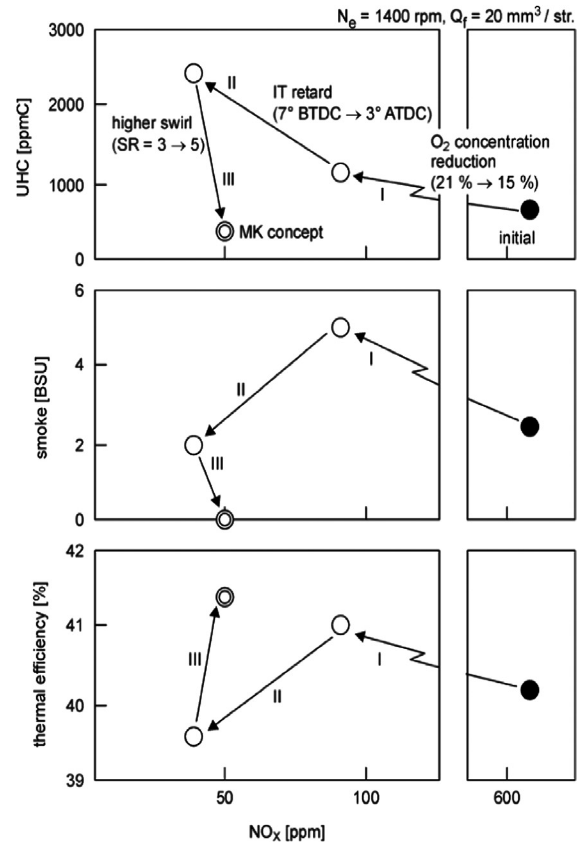

Fig. 8. Nissan MK-concept: effects of EGR, retarded injection timing (IT) and increased swirl on exhaust emissions and thermal efficiency [101].

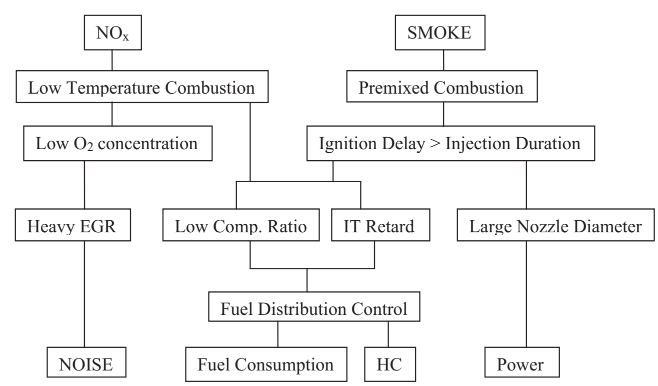

Fig. 9. Schematic of the MK combustion concept [103].

combustion, the fuel–air mixture homogeneity before ignition is required in MK combustion that can be achieved by increasing the ignition delay longer and rapid mixing with a high swirl. In the MK system, there are three features; (i) late fuel injection timing starts from 7° BTDC to 3° ATDC, (ii) high levels of EGR and (iii) high swirl ratio. The formation of NO_x emissions can be suppressed by high EGR rates (reduces oxygen concentration from 21% to 15%) and low temperature combustion. The ignition delay was increased by decreasing the compression ratio to 16:1.

Kawamoto et al. [102] found a low compression ratio was effective in expanding the MK combustion region on the high-load side. The basic concept of MK combustion is explained schematically in Fig. 9 [103]. Kimura et al. [104] examined the effects of combustion chamber insulation on the heat rejection and the thermal efficiency. The combustion chamber was insulated by using a silicon nitride piston cavity that was shrink-fitted into a

titanium alloy crown. The application of heat insulation reduced the angular velocity of the flame in the combustion chamber by about 10~20%. This reduction in the angular velocity of the flame was found to be one cause of combustion deterioration when the heat insulation was applied to the combustion chamber. The main advantage of late direct injection system is the combustion control by the injection timing over the port fuel injection and the early direct injection systems.

3.3. Narrow angle direct injection NADI™

In order to prevent fuel deposition on the cold cylinder liner, the angle between the spray must be reduced. The concept of narrow angle direct-injection (NADI) was suggested by Walter and Gatellier [105–107] to keep the fuel target within the piston bowl and avoid the interaction of the spray with the liner at advanced injection timing. The results indicated that the liquid fuel impingement on the bowl wall leads to fuel film combustion which is called "pool fire". Because of the rich air–fuel mixture and low temperature on the wall surface, the pool fire results in incomplete combustion and high soot formation for all early injection cases.

A narrow fuel spray angle and a dual injection by Myung et al. examined [108] the fuel injection angle was narrowed from 156° of conventional diesel engine to 60°, while the compression ratio was reduced from 17.8:1 to 15:1 to prevent the early ignition of the mixture. The results showed that the NO_x emissions were greatly reduced as the injection timing was advanced beyond 30° BTDC and the IMEP indicated a modest decrease although the injection timing advanced to 50–60° BTDC in the case of the narrow spray angle configuration. Fig. 10 shows the narrow spray adoption in early in-cylinder direct injection. In early in-cylinder fuel injection, the spray direction adaption is important, because the volume between the injector nozzle and piston is larger.

Tiegang et al. [109], investigated the effects of two spray injection angles (i.e., 150° and 70°) on the combustion process in an HSDI optical diesel engine employing multiple injection strategies with high injection pressures (600 and 1000 bar). The premixed combustion was observed for the 150° tip with the high injection pressure, while other cases show diffusion flame combustion features. A non-luminous flame was seen for the first injection of the 150° tip, while two types of flames are seen for the first injection of the 70° tip including a non-luminous flame and a luminous film combustion flame. The flame was observed more homogeneous for the 150° tip with the higher injection pressure, namely a combustion process close to the PCCI-like combustion, with a little soot formation. More soot luminosity is observed for the 70° tip due to fuel-wall impingement. The fuel film combustion leads to the lower NO_x emissions due to its rich mixture nature. For both the injection angles and higher injection pressures results in higher NO_x emissions, because of the leaner air–fuel mixture and higher in-cylinder temperatures for the increased injection pressures.

The French Institute of Petroleum, IFP [105] has developed a combustion system that was able to reach near zero particulate and NO_x emissions, while maintaining the performance standards of the DI diesel engines. A Narrow Angle Direct Injection (NADI™) was applied to this dual fuel mode engine which applies HCCI at part load, and switches to conventional diesel combustion to reach full load requirements. At part load (including Motor Vehicle Emissions Group-MVEG-and Federal Test Procedure-FTP-cycles), the HCCI combustion mode allows near zero particulate and the NO_x emissions and maintains a very good fuel efficiency. At 1500 and 2500 rpm, NADI™ reaches 0.6 and 0.9 MPa (6 and 9 bar) of indicated mean effective pressure (IMEP) with the emissions of NO_x and particulate under 0.05 g/kWh, which are lower by 100 and 10 times respectively than a conventional diesel engine. At full load, NADI™ system is consistent with future diesel engine power density standard. Lately, IFP [110] has developed a near-zero NO_x and particulate combustion process, the NADI™ concept, a dual-mode engine application switching from a novel lean combustion process at part load to conventional diesel combustion at full load. The narrow spark cone angle injection can reduce liner wetting problem, when the fuel is injected at early CAD for the HCCI combustion.

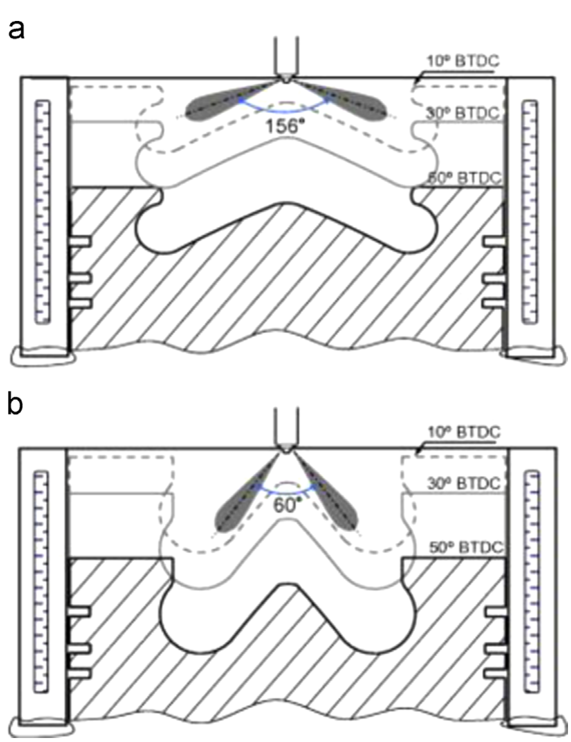

Fig. 10. Schematic diagrams of the (a) conventional diesel engine. (b) NADI™ for an early injection [108].

4. HCCI combustion control strategies

The combustion phase in HCCI engines is controlled either by altering time–temperature history or by altering the mixture reactivity [111]. Fig. 11 illustrates the methods of controlling HCCI combustion. The first group indicates the purpose of which is to alter the time–temperature history of the mixture. It includes fuel injection timing, variation of intake air temperature, variation of compression ratio (VCR) and variable valve timing (VVT). The second group attempts to control the reactivity of the charge by varying the properties of the fuel, the fuel–air ratio or the amount of oxygen by EGR. However, the homogeneous mixture preparation, before the start of ignition is the main objective of HCCI

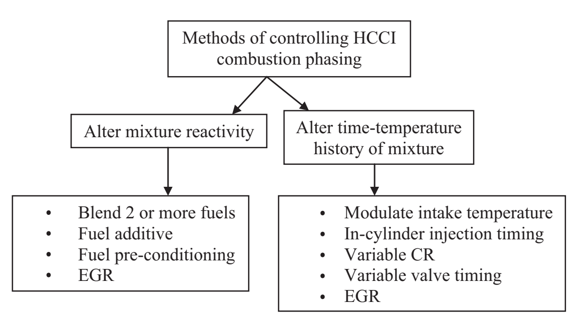

Fig. 11. Methods for controlling HCCI combustion phasing [111].

Table 5
Control strategies for HCCI combustion.

Reference[a]	Control strategy	Key features	Advantages
Su [97]	High swirl ratio	Specially designed BUMP combustion chamber used swirl ratios 3–5 with bump rings.	Improves the mixing rate of fuel and air, and high swirl is essential for the quick homogeneous mixture preparation.
Xiangang [112]	Ultra high injection pressure with small nozzle holes	Ultra-high injection pressure (300 MPa) and micro-hole nozzle (d=0.08 mm) were used.	Mixture homogeneity increases. A larger flame structure and week soot formation. There is no liquid wall wetting.
Christensen [114]	High boost pressure	Increase the indicated mean effective pressure.	Low UHC and NO_x were observed with boost pressure and engine load.
Olsson [44]	High boost pressure	Extended the high load operation.	High loads up to 16 bar brake mean effective pressure, and ultra-low emissions.
Law [117]	Variable compression ratio (VCR)	Active valve train was used to trap the exhaust gases.	Low NO_x emissions.
Christensen [17]	Variable compression ratio (VCR)	HCCI fuel flexibility was demonstrated by using primary reference fuels.	By changing the compression ratio any fuel can be burnt in HCCI engines.
Flowers [120]	Intake charge temperature	Increased the combustion efficiency by charge heating.	Emissions of hydrocarbons and carbon monoxide tend to decrease with increasing intake temperature.
Hatim [121]	Charge temperature and equivalence ratio	Start of combustion was varied with mixture reactivity.	High inlet temperature decreases ignition delay and accelerates the overall kinetics.
Martinez [157]	Heat exchanger	The intake air was heated by waste thermal energy in the exhaust gases.	Reduced the requirements of intake heating.
Gerardo [122]	Exhaust gas recirculation (EGR)	Increases ignition delay and to reduce NO_x emissions.	Longer ignition delay improved mixture homogeneity.
Fang [123]	Exhaust gas recirculation (EGR)	Variable EGR and ignition timings were used.	Lower EGR rate and delayed ignition timing should be applied at higher load and vice versa to avoid knocking.
Jan-Ola [126]	Exhaust gas recirculation (EGR)	The combustion efficiency at low load conditions was improved.	Cold EGR improved combustion efficiency of turbocharged HCCI engine at all conditions.
Mohamed [51]	Fuel additive	Di-methyl ether (DME), formaldehyde (CH_2O) and hydrogen peroxide (H_2O_2) were used as additives.	H_2O_2 addition was effective in advancing the ignition timing.
Hiraya [154]	Engine speed	Extending the engine operating regime.	As the speed increases, ignition delay becomes longer, which requires high intake temperatures. Hence, lower speeds are suitable for high load HCCI combustion.
Murase [155]	Pulsed flame jet	To enhance ignition reliability and burning rate.	Direct ignition timing control of HCCI combustion is possible with pulsed flame jet.
Hashizume [94], Yokota [95]	Split injection	The combustion process was optimised via fuel injection timing.	Split injection strategy can optimise the combustion process and control emissions.
Christensen [156]	Water injection	Controlling the auto-ignition timing.	Low NO_x emissions and delays auto-ignition timing.

[a] Only the first author is listed, all papers have multiple authors.

combustion which can be controlled (a) by increasing the degree of homogeneity; and (b) delaying auto-ignition. The important strategies for controlling HCCI combustion phase are listed in Table 5.

4.1. *Control strategies to increase the mixture homogeneity*

4.1.1. *Ultra high injection pressure with small nozzle holes*

The atomisation of the fuel inside the combustion chamber can be improved greatly by using high injection pressures (high velocity of the jet) and by decreasing the nozzle hole diameter. Xiangang et al. [112] investigated the effects of ultra-high injection pressure (300 MPa) and micro-hole nozzle (d=0.08 mm) on flame structure and soot formation of impinging diesel spray. A larger flame structure and week soot formation is detected with a micro-hole nozzle at injection pressures of 200 and 300 MPa. There is no liquid wetting for micro-hole nozzle. The fuel–air mixture homogeneity can be increased by increasing the injection pressures and by decreasing the diameter of nozzle hole.

4.1.2. *High swirl ratio*

The swirl of air in the combustion chamber also improves the mixing rate. The BUMP combustion chamber in MULINBUMP combustion system uses a high swirl ratio (3–5) with bump rings improves the mixing rate of fuel and air and high swirl is essential for the quick homogeneous mixture preparation for combustion.

4.1.3. *Pulsed fuel injection*

The early in-cylinder fuel injection requires multiple pulsed injections of the total fuel. The lower air density during early CAD injection during compression stroke causes over penetration of the fuel. The MULDIC combustion system uses a two stage fuel injection in which first injection is used for premixed lean combustion, while the second injection is meant for diffusion combustion. The multi-stage fuel injection during early compression stroke is used by the HiMICS and MULINBUMP combustion systems, where fuel–air mixing was enhanced.

4.1.4. *High boost pressure*

Supercharging or turbocharging is used in HCCI engines to extend the domain of operation [20,113]. Supercharging in HCCI combustion increases the Indicated Mean Effective Pressure (IMEP) to 14 bar [114]. Supercharging has the capacity to deliver increased charge density and pressure at all engine speeds while turbocharging depends upon the speed of the engine. The in-cylinder density and volumetric efficiency can be improved with a high boost pressure. The evaporation of the fuel is increased with a high intake pressure due to high in-cylinder temperatures. The mixing time can be decreased with the boost pressure is advantageous with all early injection systems. The combustion efficiency can be improved slightly at high boost levels, and cooled EGR rates was introduced [115]. But, Taewon et al. [116] found, that the increase of intake boost pressure shortened ignition delay which is not favourable for the MK combustion. Olsson et al. [44] extended the operating range of 6-cylinder truck engine modified to turbocharged HCCI engine. This study proves the possibility to

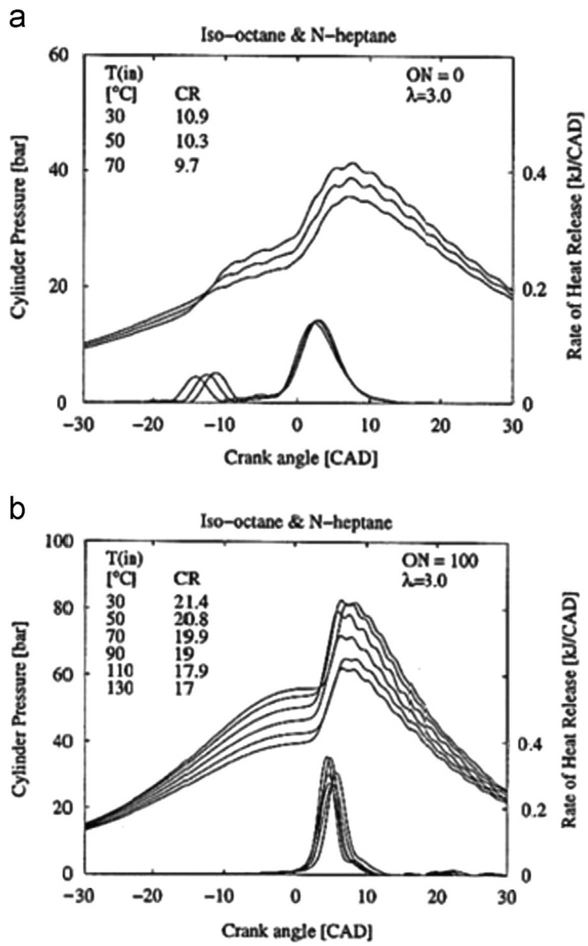

Fig. 12. Cylinder pressure and heat release for (a) PRF (n-heptane) (b) PRF (iso-octane) [17].

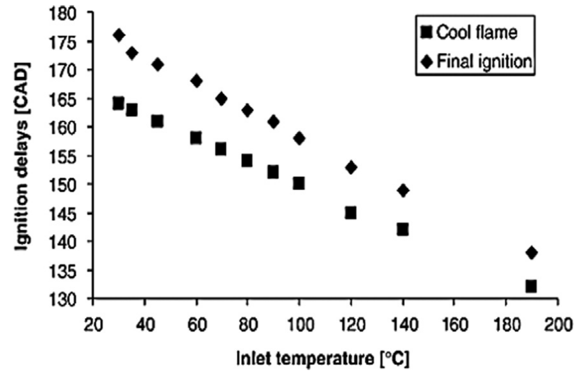

Fig. 13. Ignition delays as a function of the inlet temperature for n-heptane [121].

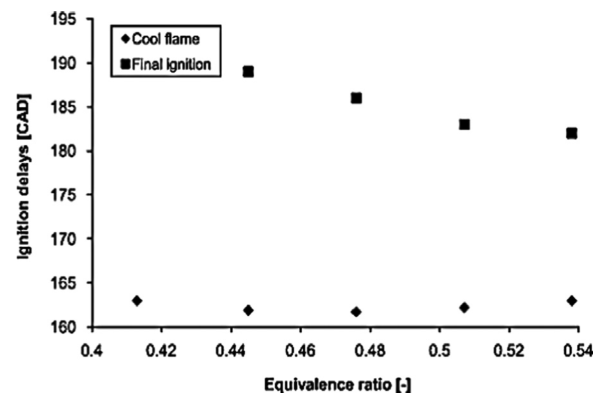

Fig. 14. Ignition delays as a function of the equivalence ratio for iso-octane [121].

achieve high loads, up to 16 bar Brake Mean Effective Pressure (BMEP), and ultra-low NO_x emissions.

4.2. Control strategies to delay the auto-ignition

4.2.1. Variable compression ratio (VCR)

The start of ignition (SOI) can be delayed by decreasing the compression ratio of the diesel engine, but it should not be decreased to much as it suffers from the thermal efficiency. Many researchers worked on VCR engines to delay auto-ignition of the fuel and to utilise many alternative fuels in HCCI engines. To achieve HCCI combustion decreasing inlet temperatures and lambdas, higher compression ratios were need to maintain correct maximum brake torque and concluded that VCR can be used instead of inlet heating [117]. Christensen et al. [17] demonstrated a low octane (n-heptane) fuel or a high octane (iso-octane) fuel or a medium octane fuel which is the most suitable for the HCCI operation, regarding fuel efficiency and emissions. The compression ratio was changed from 10:1 to 28:1, shows fuel flexibility of HCCI engine using VCR. Fig. 12(a and b) shows the cylinder pressure and the heat release for zero octane and 100 octane fuels. By changing the compression ratio any fuel can be burnt successfully in an HCCI engine.

4.2.2. Charge temperature and equivalence ratio

The auto-ignition of fuel–air mixture is a very sensitive to intake air temperature changes, as small as 5–10 °C [118,119]. Hence, the combustion control is very difficult task in order to achieve high efficiency without any knock. Diesel fuel doesn't require any charge heating, as it can be burnt easily with a compression ratio of 16. For low cetane fuels, modulate intake air temperature is necessary to reach its auto-ignition temperature near the TDC for combustion. A higher intake temperature advances combustion but the engine volumetric and thermal efficiency can be largely reduced, due to the fact that, if ignition is advanced into the compression stroke, it will cause significant negative work on the piston. Flowers et al. [120] studied cylinder-to-cylinder effects on the variable intake temperature and propane fuel flow rate. Hatim et al. [121] analysed of the influence of the inlet temperature, equivalence ratio and compression ratio on the HCCI auto-ignition process of primary reference fuels. Figs. 13 and 14 shows the ignition delays as a function of the inlet temperature and equivalence ratio respectively for primary reference fuels.

4.2.3. Exhaust gas recirculation (EGR)

The technology of EGR is widely used in HCCI combustion due to its high potential of controlling the auto-ignition of time–temperature history and enhancement of NO_x emission reduction. The EGR can be categorised into internal and external EGR. Internal EGR is acquired by the exhaust gas trap (EGT) using the negative valve overlap (NVO) and variable valve timing (VVT) methods. The most practical means to delay the auto-ignition in an HCCI engine is through the addition of high levels of EGR into the intake. The inert gases present in the EGR can be used to control the heat release rate due to its impact on chemical reaction rate, which can delay the auto-ignition timing. Hence, EGR reduces the heat release rate, and thus lowers the peak cylinder temperature due to the constituents of EGR (mainly CO_2 and H_2O) having higher specific heat capacities.

The MK combustion system uses a high EGR to reduce the NO_x emissions up to 98% less than conventional diesel engine. Gerardo

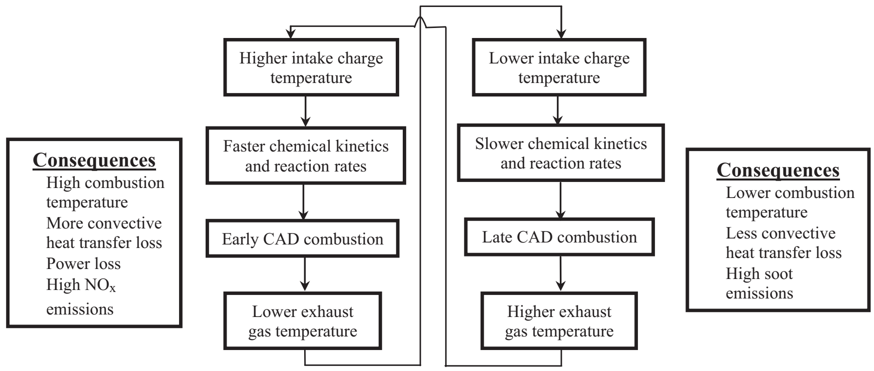

Fig. 15. Consequences of intake charge temperature and EGR in HCCI engine.

et al. [122] studied the effect of EGR on ignition delay and emissions are (i) reducing oxygen concentration at the intake, (ii) lowers the adiabatic flame temperature, (iii) increases the ignition delay with a slower pressure rise rate that also improves the combustion noise, (iv) decreases NO_x emissions, and (v) a high EGR rate decreased volumetric efficiency and increased smoke emissions. Ganesh et al. [67] reported that the EGR has two primary effects on HC emission (a) the intake of some un-burnt HC with exhausted gas into the next cycle leads to a decrease in HC emissions, and (b) the decrease of combustion temperature in the cylinder leads to an increase in HC emissions. Fang et al. [123] investigated on the dual effects of EGR on the BSFC and reported that, the EGR can reduce the pumping loss, and it will lead to slowing the flame propagation speed, making the combustion process far from TDC and decreasing work efficiency of combustion in cylinder. Fig. 15 shows the consequences of intake charge temperature on EGR addition.

The combustion limit towards leaner air–fuel mixture and the tolerance to the EGR can be significantly extended. The low heating value of lean mixtures and the high heat capacity of EGR can lower the peak temperature of combustion, thus reduce NO_x emission. Up to 95% reduction in NO_x emission has been obtained experimentally [124,125]. At low load, the combustion efficiency is the most important one in HCCI combustion which is improved by EGR [126]. The EGR also used to control the HCCI auto-ignition. The hot EGR advances the combustion timing while cold EGR retards the combustion timing.

4.2.4. Fuel modification

The auto-ignition characteristics of the fuel–air mixture can be controlled with fuel blending or/and additives. For HCCI combustion volatility and auto-ignition characteristics of the fuel are important [127]. Fuel requirements for HCCI engine operation by Rayn et al. [128] on constant volume combustion bomb experiments shows the primary properties of fuel relate to the distillation characteristics and the ignition characteristics. Research octane number (RON) is a measure of fuel resistance to knock while motor octane number (MON) is a measure of how the fuel behaves when under load. Kalghatgi [129,130] developed an Octane Index (OI) (function of MON and RON) for measuring the auto-ignition or anti-knock quality of a practical fuel at different operating conditions. Kalghatgi's lower OI shows earlier combustion phasing. Shibata et al. [131,132] showed a relationship between RON and low temperature heat release (LTHR) which has a strong impact on high temperature heat release (HTHR). He studied 12 hydrocarbon constituents for HCCI combustion in which olefins and aromatics (except benzene) have a function to retard combustion phasing (LTHR) while iso-paraffins and n-paraffins have a function to advance combustion (HTHR). The effects of cetane number (CN) on HCCI performance, auto-ignition, and emissions were investigated by some researchers [133–137]. Aroonsrisopon et al. [138] found that HCCI combustion is a strong function of fuel composition and cannot be predicted by octane number. Shibata et al. [131] demonstrated that the fuel chemistry directly affected by LTHR and the subsequent main combustion stage (HTHR). Bunting et al. [139,140] found high cetane fuels have stronger LTHR behaviour and does not require high intake temperature for auto-ignition. He concluded that low Cetane fuels are more desirable for pure HCCI combustion [141]. The HCCI combustion of high octane fuel (iso-octane) shows single stage ignition while fuel blends shows two-stage ignition [42]. Fig. 16 shows the single and two-stage heat release rates for iso-octane and PRF80.

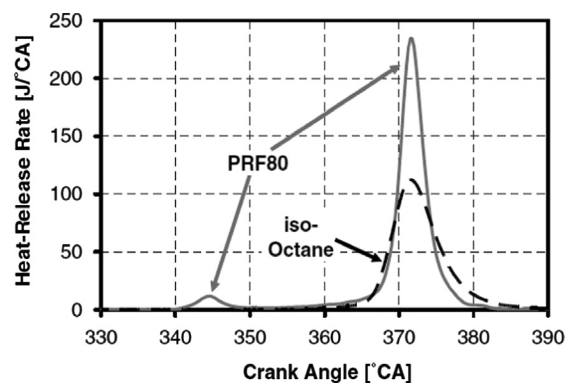

Fig. 16. Heat-release traces for iso-octane and PRF80 [42].

The HCCI auto-ignition can also be controlled with fuel additives. An additive fuel in HCCI combustion has the ability to promote or inhibit the kinetics of auto-ignition chemistry. Dimethyl ether (DME) has good characteristics of auto-ignition and combustion with low flame temperature, and exhibit two-stage heat release [142]. Mosbach et al. [143] used numerical simulation of ethanol and diethyl ether (DEE) by means of a stochastic reactor model. Mohamed et al. [51] studied the effect of additives such as di-methyl ether (DME), formaldehyde (CH_2O) and hydrogen peroxide (H_2O_2) for the control of ignition in natural gas HCCI engines. The study revealed that high octane fuels require auto-ignition promoters in HCCI combustion engines such as natural gas [144–146], LPG [147–149] and ethanol [150]. The fuel inhibitors extend the ignition delay and promote low temperature oxidation [151,152]. Clothier et al. [158] studied the effect of fuel additives during cold start conditions. He found the addition of DEE to diesel fuel significantly inhibits its ignition. The fuel additives in HCCI engines control the start of ignition or extend the maximum load limit.

5. Potential future research directions

Globally, there are many research and development activities are going on, in the area of HCCI combustion, due to their potential to meet emission standards as well as high thermal efficiency. However, in order to realise them in modern engines, there are still many challenges to be solved. For diesel-fueled HCCI combustion will need to focus on three different methods (i) introducing the fuel, (ii) fuel–air mixing, and (iii) ignition timing control. For example, at low ambient conditions; the lower vaporisation of diesel fuel causes difficulties in creating homogeneous mixture before start of ignition. In addition, wall wetting of fuel if the spray over penetrates resulting low combustion efficiency and high UHC emissions.

In reality, the HCCI combustion systems need further development of three main technologies. They are (i) flexible fuel injection systems which make the homogeneous mixture before auto-ignition of charge, (ii) EGR control system to control combustion and emissions, and (iii) closed-loop feedback system. Despite these technical problems, many efforts made to develop diesel-fueled HCCI systems gave a high degree of success and introduced into the commercial market. These include the Toyota UNIBUS system, which uses the early in-cylinder injection strategy, and the Nissan MK combustion system, which uses the late in-cylinder injection strategy. However, both these systems presently operate in the HCCI mode at part load only. For short term automotive applications, “dual mode” operations are best suited. The expansion of the operating range from part load to full load is promising research.

The utilisation of potential alternative fuels and fuel additives in an HCCI engine is to widen the operating regime. Neat alternative fuels, both methanol and ethanol have demonstrated in widening the HCCI operating regime and compared to gasoline and PRF fuels. The efforts are being made to use neat ethanol in the commercial HCCI engine market show promise.

Fuel additives have a strong potential to promote auto-ignition for HCCI operation at light loads. A systematic evaluation of ignition improvers to promote expansion of operating regime should be performed. Some other issues like fuel handling and thermal stability of the additives need to be solved before they represent in a viable technology. Addition of DEE to diesel fuel significantly inhibits its ignition and its performance in HCCI combustion has not yet been determined. The reduction of UHC emissions at high loads and CO emissions at low loads needed research. The interesting research needed the development of low temperature oxidation catalyst for HCCI engines at light load.

Through this review, the authors conclude that the high efficient and clean combustion system design will be the most interesting topics in the future. The research on HCCI engines continues in two directions simultaneously, until the zero emission vehicles coupled with energy efficient HCCI combustion systems are developed.

6. Conclusions

The conclusions of the review on the mixture formation and control strategies adopted in the HCCI mode diesel engines are as follows:

- The HCCI combustion engines have the potential to improve the thermal efficiency, while reducing the trade-off emissions in conventional diesel engines.
- The homogeneous mixture preparation and auto-ignition control are the main targets of researchers in HCCI combustion. The elimination of local fuel-rich zones and soot emissions is possible only by effective homogeneous charge preparation, while controlling auto-ignition is to achieve higher thermal efficiencies.
- LTC is widely implemented in all power generation systems, as it offers many advantages on both efficiency and emissions. HCCI combustion comes under the LTC through the utilisation of homogeneous charge for combustion initiated by auto-ignition.
- The new combustion methods in which diesel-fueled HCCI combustion are PREDIC, MULDIC, MK etc. show higher efficiencies and lower emissions.
- The port fuel injection has a high degree of mixture homogeneity compared to other injection methods, but lacks start of combustion control.
- The advantages in MK combustion concept are combustion control, without any fuel wall impingements and zero soot emissions.
- The emissions of NO_x and smoke are low in all advanced combustion modes in comparison with conventional diesel engine, while the UHC/CO emissions are increasing in all low temperature combustion concepts except in MK combustion.
- The mixture homogeneity with a multi-pulse fuel injection in NADI™ system is promising for early in-cylinder injection for attaining a high degree of mixture homogeneity with minimum wall wetting.
- The soot formation can be decreased by using ultra high pressure injection system with small nozzle holes.
- The pulsed injection system and swirl ratio improves the mixture homogeneity, while the boost pressure improves the fuel evaporation and volumetric efficiency.
- The combination of equivalence ratio and charge temperature decides the auto-ignition characteristics of the fuel for HCCI combustion.
- The EGR can extend the auto-ignition, but combustion control by slowing the heat release rate is possible.

  The VCR is used to change the auto-ignition for flexible fuels which can be altered by fuel blending or additives also.

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